Desiccant Dehumidification with Hydronic Radiant Cooling System for Air-Conditioning Applications in Humid Tropical Climates

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ABSTRACT

This paper discusses the feasibility of a hybrid desiccant dehumidification system combined with chilled ceiling for air-conditioning applications in humid tropical climates. The study presents a design/operation guide of the hybrid system. The study also indicates definite merit of the hybrid system when the ventilation air requirement of the conventional system is above a certain threshold. This is particularly so in many practical applications, where a high ventilation air requirement is desirable or mandated, such as operating theaters and certain hospital wards. A trial run on the facility indicates the viability of the scheme, particularly the absence of condensation/sweating of chilled panels. In the same context the facility developed to conduct experiments is described. For a space loading of 0.1 kW/m² (31.71 Btu/h·ft²), any ventilation rate above 2% for a conventional system offers opportunity for downsizing chiller capacity of the hybrid system. Based on an indicative energy analysis, the proposed hybrid system becomes more energy efficient than a conventional system when the required ventilation rate is 30% and above.

INTRODUCTION

The conventional vapor compression refrigeration cycles used in commercial air conditioners are energy intensive, while evaporative, desiccant, and solar coolers are not economically viable as stand-alone systems. Evaporative cooling is a fairly attractive option for comfort applications in arid tropical climates where reasonable cooling is achieved economically. However, cooling is accompanied by relatively high humidity, that may not always be acceptable. Desiccant cooling is gaining acceptance as an alternative means of cooling (Dhar and Singh 2001; Jain et al. 2000; Kini et al. 1990), but large-scale use of the same is limited because of the inherent problem of the need for precooling of desiccated air and effective and economical desorption of desiccant. One positive aspect, however, is the opportunity to use low-grade energy, e.g., solar energy, natural gas, bio mass, etc. Use of solar energy is desirable, but the insolation intensity varies at different times and geographical locations and its availability is not continuous. Another positive feature of the desiccant system is the likely reduction of the use of ozone-depleting HCFC products. Control of humidity can be achieved better than with conventional systems employing vapor compression systems, since sensible and latent cooling are decoupled and they can be controlled separately. Better indoor air quality can be maintained for desiccant systems because the fresh air supply percentage is very high (usually 100%). Desiccant systems also have the capability of removing airborne pollutants. The use of a hybrid desiccant air conditioner, where desiccant is used to adsorb atmospheric moisture complemented by a conventional refrigeration unit (in the present case, a chilled ceiling panel) providing cooling, is a proposition that merits serious consideration.

Hydronic radiant cooling (HRC) provided by a chilled ceiling (CC) combined with desiccant dehumidification (DD) is a relatively new concept. In recent years both simulation studies and experimental research on HRC and displacement ventilation (DV) have been reported (Alamdari et al. 1998; Loveday et al. 2002; Mumma 2001; Novoselac and Srebric 2002; Rees and Haves 2001). For the combination of HRC and DD some simulation studies have been published (Niu et al. 1995; Zhang and Niu 2003a). However, for the combination of HRC and DD, no experimental work has been reported to date. The inherent advantage of the system is that the chilled ceiling

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temperature does not have to be lower than dew-point temperature, resulting in potential downsizing of the refrigeration system used. Another advantage claimed for radiant cooling is that cooling would be provided directly and more evenly to the occupants without causing draft, resulting in better thermal comfort (Feustel and Stetiu 1995). Although chilled ceilings have been used in European countries, their use in humid tropical climates is faced with two daunting challenges—first, the fact that 100% cooling capacity cannot be met and, second, the ever-present condensation problem. Of necessity, therefore, there is a need to decouple the space sensible and latent heat (Mumma 2002; Niu et al. 2002). There is thus a case for investigating the feasibility of a hybrid air conditioner comprising a chilled ceiling that would provide hydronic radiant cooling and a desiccant dehumidifier supplying dehumidified and precooled air. The current project was thus conceived to carry out a design study to establish the viability of such a hybrid system followed up by subsequent experimental verification. In the same context, the paper discusses the facility developed wherein the experiments are planned to be conducted.

EXPERIMENTAL TEST FACILITY

With the above objective, a facility has been developed at the Fluid Mechanics Laboratory of the Universiti Sains Malaysia. In addition to carrying out the above-mentioned research, the facility has been designed to accommodate multi-disciplinary projects in diverse areas, including experimental verification of CFD simulation and research in the area of thermal comfort.

The experimental facility comprises (a) a chilled water circuit, (b) chilled ceiling, (c) desiccated air displacement ventilation (DADV) system, and (d) climate chamber. Watt-hour meters have been installed for recording the energy consumption of the different devices.

Chilled Water Circuit

The chilled water circuit is made up of an air-cooled chiller with nominal cooling capacity of 11.72 kW (39,988 Btu/h) (3.3 TR) providing chilled water to the chilled ceiling panel and air cooler downstream of the desiccant dehumidifier. A three-way bypass valve has been installed in the chilled water circuit to control the ceiling panel temperature by means of a thermostat. A bypass line supplies chilled water to the precooler to bring down the temperature of the dehumidified air. Figure 1 shows a schematic diagram of the combined DADV and CC system.

The various processes of the system are represented on a skeleton psychrometric chart shown in Figure 2. Outdoor air at ambient state 1 is dehumidified and heated to state 2 as it passes through the rotary desiccant wheel. This dehumidified air is then cooled first by a heat exchanger (yet to be installed) to point 3 followed by further cooling to state point 4 by the water precooler (optional) and to state point 5 by a chilled water precooler. This air is then delivered into the climate chamber, resulting in the condition represented by point 6.

Chilled Ceiling

There are two practices in chilled ceiling construction—one is a drop ceiling, or T grid type, and the other is the hanging element type (Mumma 2001). In such systems, chilled water is made to flow through the tubes embedded in the ceiling panels, typically maintaining ceiling surface temperature in the range of 16°C to 19°C (60.8°F to 66.2°F). Chilled ceilings can remove thermal loads up to 100 W/m^2 (31.71 Btu/h·ft^2) of floor

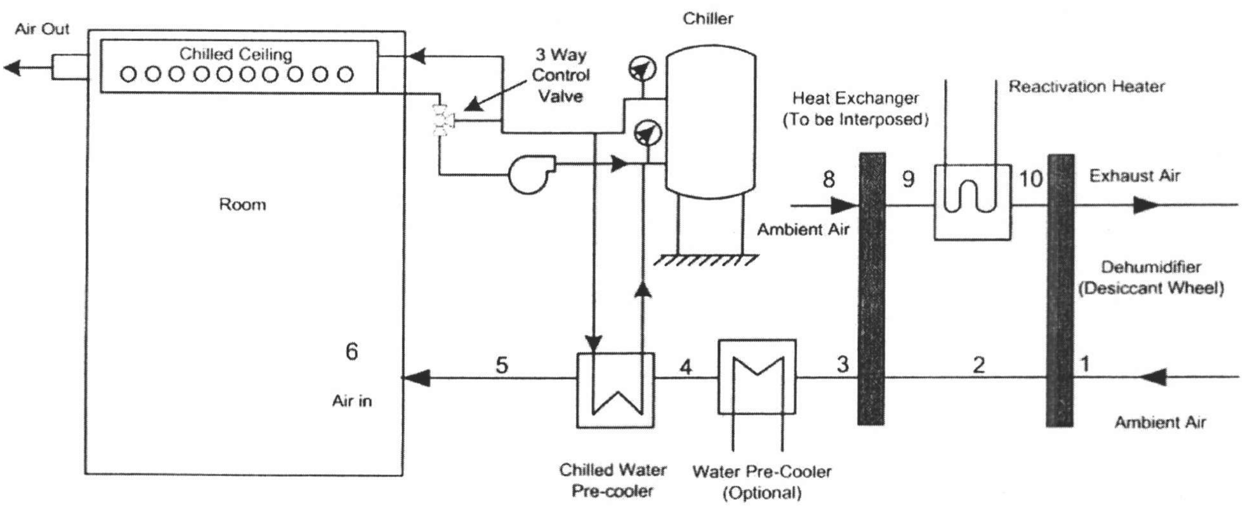

Figure 1 Schematic diagram of the combined DADV and CC system.

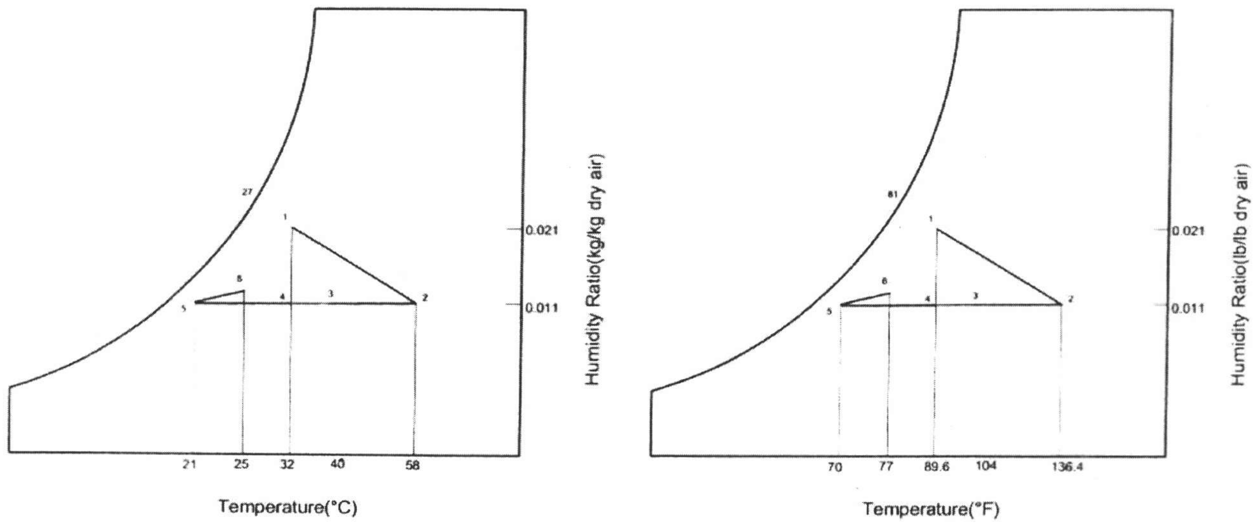

Figure 2 Representation of the cycle on a skeleton psychrometric chart.

area by the combined processes of radiation and convection (Loveday et al. 1998). This system is considered to enhance the thermal comfort sensation of occupants. When combined with displacement ventilation, the advantages offered by each system separately, i.e., improved air quality and enhanced thermal comfort, can be harnessed.

In the present facility, a drop-down type ceiling has been used, as shown in Figure 3. The custom-built chilled ceiling comprises 12 flat panels made of aluminium plates of 1 mm (0.039 in.) thickness occupying 70% of the total ceiling area. The copper cooling tubes used are of 12 mm (0.468 in.) diameter with 150 mm (5.85 in.) spacing between the tubes. There are two headers providing chilled water to the individual panels through flexible tubes (Figure 4). Provisions have been made to isolate chilled water flow through specific panels.

Environmental Chamber

A climate chamber has been built in which the chilled ceiling (CC) has been installed for conducting this research. The 4.25 m × 3.75 m × 3 m (13.94 ft × 12.3 ft × 9.84 ft) chamber has been constructed with demountable clip-lock type insulated panels. The 100 mm (3.9 in.) insulated panels are of galvanized steel sheets laminated to an insulation core of polyurethane. A desiccant-based air conditioner supplies dehumidified air to the CC chamber. Figure 5 shows external views of the environmental chamber.

Desiccant Dehumidification System

Use of chilled ceiling systems in hot, humid regions poses the problem of water condensation on ceiling surfaces. It is, therefore, essential to use an independent and complementary air dehumidification system. Among the various options, desiccant dehumidification is the most appropriate one. A commercial silica gel desiccant wheel of the fluted flat bed type has been installed to supply dry air to the CC chamber. The dehumidifier is of the rotary type, which dries air by the process of continuous physical adsorption. The moisture is adsorbed in the dehumidification sector by slowly rotating the fluted, metal silicate desiccant synthesized rotor and is exhausted in the reactivation sector by a stream of hot air in counterflow. Following the reactivation process, the adsorption sector is again ready to adsorb the moisture. Thus, the two processes of moisture adsorption and reactivation take place with separate airflows continuously and simultaneously. Trial run measurements were (a) process inlet condition, 32°C (89.6°F) dry-bulb temperature and 27°C (80.6°F) wet-bulb temperature, and (b) the process outlet condition, 58°C (136.4°F) dry-bulb temperature and 28°C (82.4°F) wet-bulb temperature. Figure 6 shows the air dehumidification and regeneration processes through the desiccant wheel.

Data Acquisition System

A comprehensive data acquisition (DAQ) system has been developed for automatic recording of temperature, mean radiant temperature, relative humidity, and velocity at various locations in the climate chamber. The DAQ hardware comprises a Pentium processor-based desktop computer, a data logger, and a data acquisition software. Shielded thermocouples are used to record simultaneously temperatures at eight points of the strategic grid in the chamber. With automatic temperature data logging into the computer, both the tedious work of reading data as well as the differential in timing to read the data would be eliminated. Furthermore, with the absence of a human (heat source) inside the chamber, the

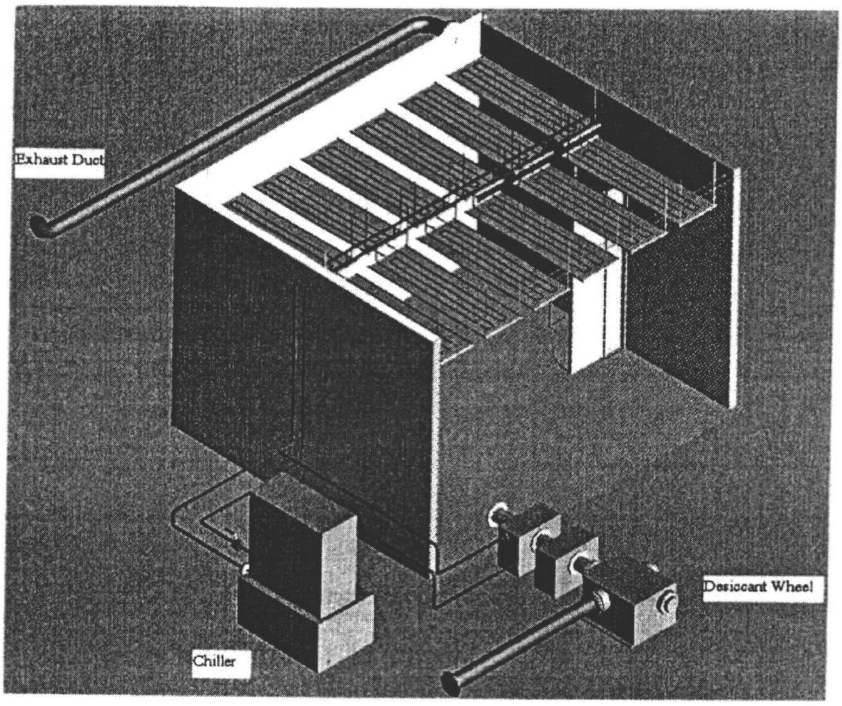

Figure 3 Three-dimensional view of the experimental facility.

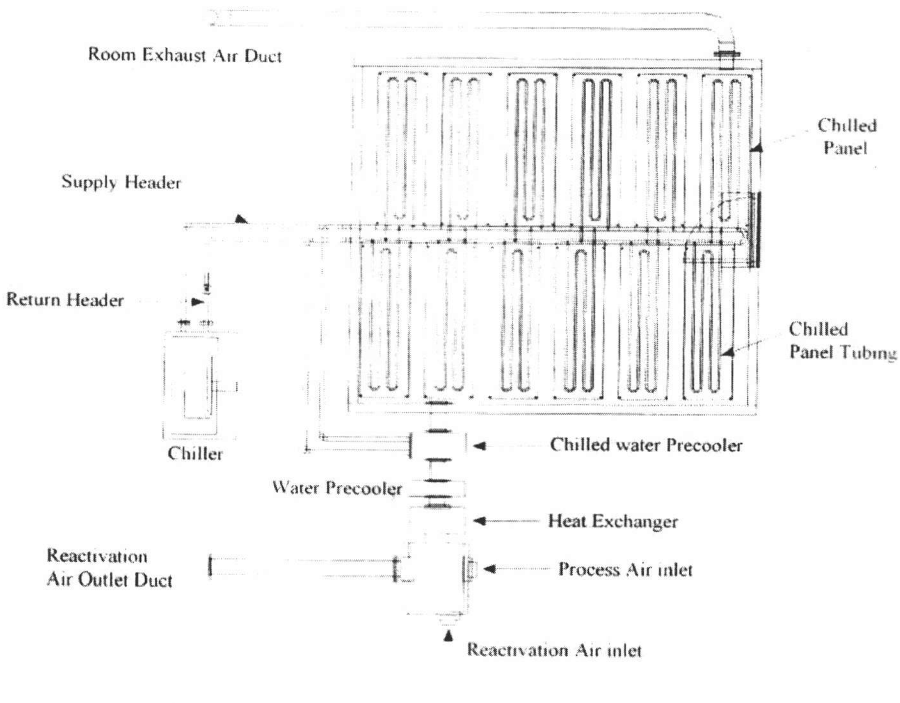

Figure 4 Chilled water and air circuit.

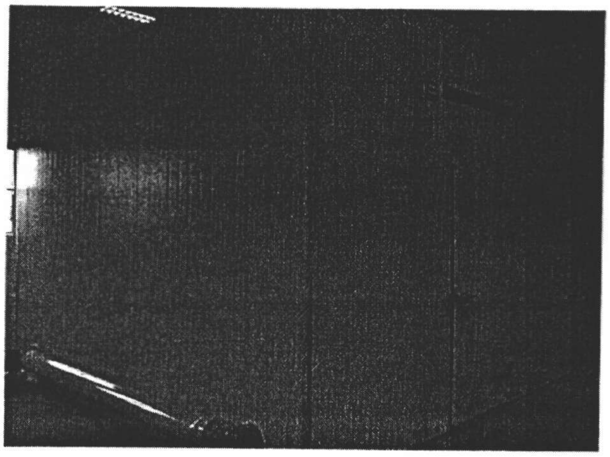

Figure 5 External views of the environmental chamber.

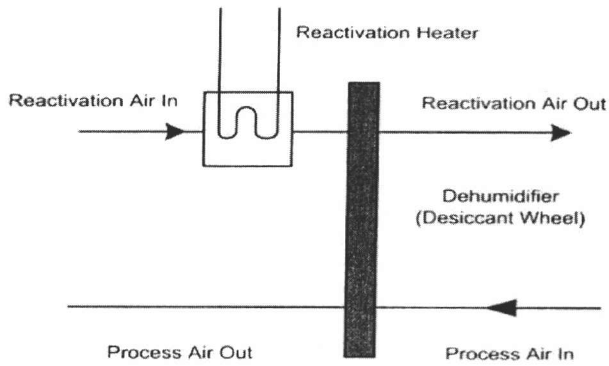

Figure 6 Air dehumidification and regeneration process.

data collected would, therefore, be less erroneous. The data acquisition arrangement is shown in Figure 7.

DESIGN AND FEASIBILITY STUDY

As mentioned earlier, the objective of the study is to establish if the system is practical and economical vis-à-vis a conventional mode of air conditioning employing a vapor compression cycle. More specifically, there is a need to optimize the critical parameters of the hybrid system, e.g., ceiling temperature, ventilation air temperatures (dry and wet bulb), and the supply volume flow rate in relation to the space cooling load and comfort criteria. Following the same study, a preliminary analysis has been carried out to get some idea about the effect/impact on chiller sizing and indicative energy implications.

Panel Heat Transfer

The chilled ceiling panels remove the sensible heat from the space by a combination of convection and radiation. The radiant heat transfer is governed by the Stefan-Boltzmann equation. In practice, for most building enclosures the thermal emittance is 0.9, and for this thermal emittance, the radiation view factor becomes 0.87. When these common values are placed into the Stefan-Boltzmann equation, the following equation (ASHRAE 1996, page 6.2, Equation 5) emerges:

$$q_r = 5 \times 10^{-8}[(t_p + 273)^4 - (AUST + 273)^4] \qquad (1)$$

where

q_r = radiant heat transfer,

t_p = effective panel surface temperature, and

$AUST$ = area-weighted average temperature of the nonradiant panel surfaces of the room.

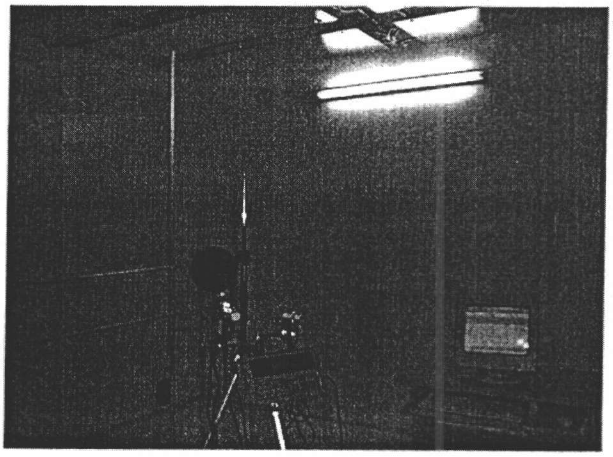

Figure 7 Computerized data acquisition (DAQ) instruments.

The convection coefficient is defined as the heat transferred by convection between the air and the panel. The rate of heat transfer by convection is a combination of natural and forced convection. Convection in a panel system is usually natural. In natural convection, air motion is generated by the cooling of the boundary layer of air and being displaced by the warmer air in the room. Research suggests that for practical panel cooling applications without forced convection, the natural convection heat transfer for cooling is given by the following equation (ASHRAE 1996, page 6.3, Equation 10):

$$q_c = 2.12|t_p - t_a|^{0.31} - (t_p - t_a) \qquad (2)$$

where

q_c = convective heat transfer and

t_a = room air temperature.

Design Analysis

The analysis is based on a specific system where a chilled ceiling has been considered in the installed climate chamber of 47.81 m³ (1688.65 ft³) volume. The inside conditions considered are 25°C (77 °F) and 50% RH, while the outdoor design conditions considered are 34°C (93.2°F) DB and 28°C (82.4°F) WB. A desiccant-based air dehumidifier supplies dehumidified air to the climate chamber, while the chiller supplies chilled water to chilled ceiling panels (Figure 1) that occupy 100% of the total ceiling area. Ceiling temperature is maintained within a range of 15°C to 18°C (59°F to 64.4°F) by a thermostat controlling a three-way bypass valve in the chilled water line. Chilled water is also tapped to cool air in the heat exchanger downstream of the desiccant wheel. An additional water precooler is interposed between the desiccant wheel and the chilled water precooler to remove the greater part of the heat of condensation. The room cooling load is removed partially by the chilled ceiling and the balance by the desiccated and cooled ventilation air. Temperature and volume of the supply air and ceiling temperature are varied to ensure a comfortable environment. The regeneration of the desiccant is done by ambient air heated by a reactivation heater, to be substituted later by gas and solar heaters.

The analysis is based on a space loading of 0.1 kW/m² (31.71 Btu/h·ft²) and sensible heat ratio (SHR) of 0.7, which are representative of hot and humid climates. The simulation study has been carried out to determine the required supply air temperature for a range of chilled ceiling temperatures and supply air volumes. The hybrid system load is made up of the (a) radiant cooling load, (b) convective cooling load, and (c) displacement ventilation load. The radiant cooling load and the convective cooling load were obtained by using, respectively, Equations 1 and 2. The results are tabulated in Table 1.

Table 1. Hybrid System Performance Analysis (SI)

Chilled Panel Temperature °C	Heat Removal by Chilled Ceiling: Radiation W/m²	Heat Removal by Chilled Ceiling: Convection W/m²	Heat Removed by Displacement Ventilation W/m²	Inlet Volume Flow Rate m³/h	Supply Temp °C	Displacement Ventilation Load (kW)	Total Load (kW)
15	46.5	43.28	10.22	67*	17	0.33	1.76
				100	20	0.40	1.83
				150	21.7	0.51	1.99
				200	22.54	0.68	2.11
				250	23	0.85	2.28
16	41.5	37.7	20.8	67	10		
				100	15	0.31	1.85
				150	18.34	0.70	1.96
				200	20	0.81	2.07
				250	21	0.93	2.19
17	36.5	32.31	31.19	67	-2.68		
				100	10.01		
				150	15	0.88	1.97
				200	17.52	0.935	2.02
				250	19	1.09	2.18
18	31.5	27	41.4	67	-4.7		
				100	5		
				150	12		
				200	15		
				250	17	1.24	2.17

* ASHRAE ventilation standard

Table 1. Hybrid System Performance Analysis (I-P)

Chilled Panel Temperature °F	Heat Removal by Chilled Ceiling Radiation Btu/h·ft^2	Heat Removal by Chilled Ceiling Convection Btu/h·ft^2	Heat Removed by Displacement Ventilation Btu/h·ft^2	Inlet Volume Flow Rate ft^3/h	Supply Temp °F	Displacement Ventilation Load (Btu/h)	Total Load (Btu/h)
59	14.74	13.72	3.24	2366.44*	62.6	1126	6005
				3532	68	1365	6244
				5298	71.06	1740	6790
				7064	72.57	2320	7199
				8830	73.4	2900	7779
60.8	13.15	11.95	6.59	2366.44	50		
				3532	59	1058	6312
				5298	65	2388	6687
				7064	68	2764	7063
				8830	70	3173	7472
62.6	11.57	10.24	9.89	2366.44	27.17		
				3532	50.01		
				5298	59	3003	6722
				7064	63.53	3190	6892
				8830	66.2	3719	7438
64.4	99.8	8.56	13.12	2366.44	23.54		
				3532	41		
				5298	53.6		
				7064	59		
				8830	62.6	4231	7404

* ASHRAE ventilation standard

Following the same exercise, Figure 8 is plotted showing the variation of supply air temperature versus required supply air volume for chilled panel temperatures ranging from 15°C to 18°C. It shows that for a low supply volume of 67 m^3/h (2,366 ft^3/h), the required supply air temperature for panel temperatures of 17°C (62.6°F) and 18°C (64.4°F) is too low (−2.68°C [27.17°F] and −4.7°C [23.54°F], respectively) to be economical. From the same analysis, a panel temperature of 15°C, however, appears feasible due to the fact that supply air temperature ranges between 17°C and 23°C (62.6°F and 73.4°F). However, with a higher supply volume of 150 m^3/h, even a panel temperature of 18°C (64.4°F) is marginally practicable as the supply air temperature is 12°C (53.6°F), which is about the same as supply temperature for conventional systems. The exercise may be viewed in the context that for a conventional all-air system, the supply air volume would be 275 m^3/h (9,713 ft^3/h) based on design room temperature of 25°C (77°F), supply air temperature of 13°C (55.4°F), and coil bypass factor of 15%.

The performance of this hybrid system needs to be compared to that of a conventional system with recirculation mode of air conditioning (Figure 9a). Identical space loading (0.1 kW/m^2 [31.71 Btu/h. ft^2]) and design conditions (25°C [77°F] and 50% RH) have been considered for both systems. The various processes have been represented on a skeleton psychrometric chart shown in Figure 9b. Outdoor air at state 2 is mixed with return air from the conditioned space at state 1 to give state 3. This air is dehumidified and cooled to state 4 as it passes through the evaporator. The same analysis has been repeated for a range of ventilation air supply (10% to 100%) and tabulated in Table 2.

Figure 10 has been plotted showing the chiller load of a conventional system for different ventilation rates (also expressed in percentage of total air supply). Across the same curve, two horizontal lines have been drawn through A and B, which represent the minimum and maximum total load for the hybrid cycle (obtained from Table 1). Point A represents 15°C (59°F) chilled panel temperature and 67 m^3/h (2,366 ft^3/h) of airflow, with corresponding total load of 1.76 kW (6005 Btu/h). Point B represents 15°C (59°F) chilled panel temperature and 250 m^3/h (8,830 ft^3/h) of airflow where the corresponding load is 2.28 kW (7779 Btu/h). This graph can be used to determine

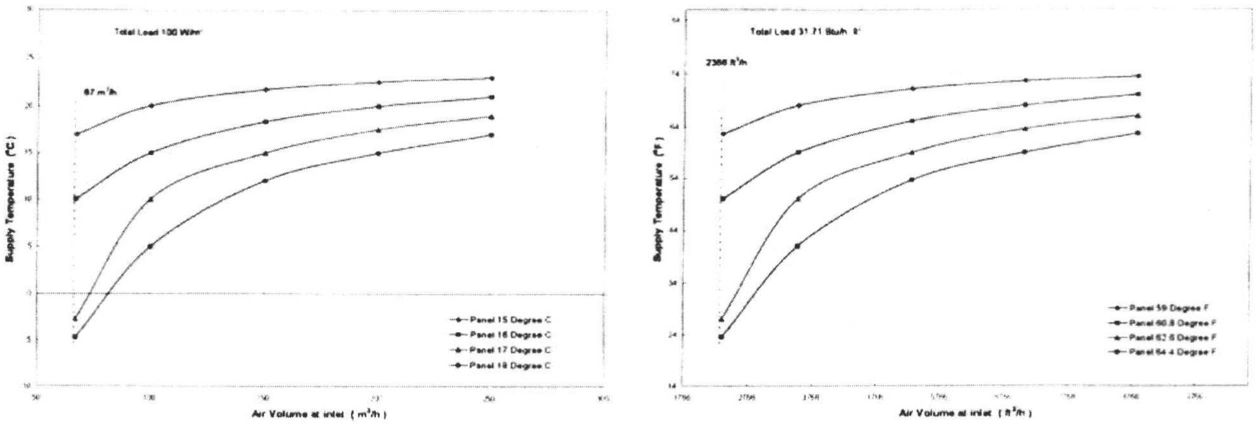

Figure 8 Required displacement volume flow rate for different supply and panel temperatures.

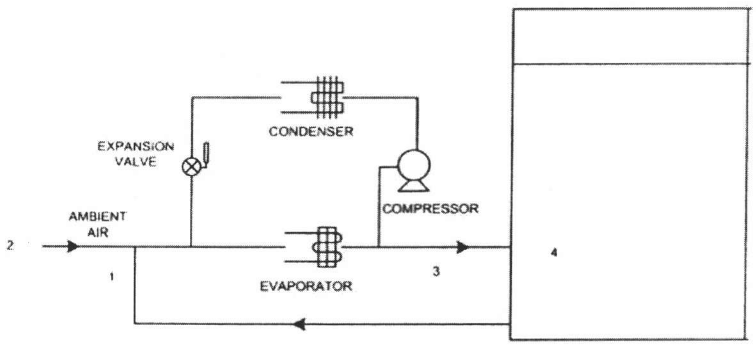

Figure 9a Schematic diagram of the conventional all-air system.

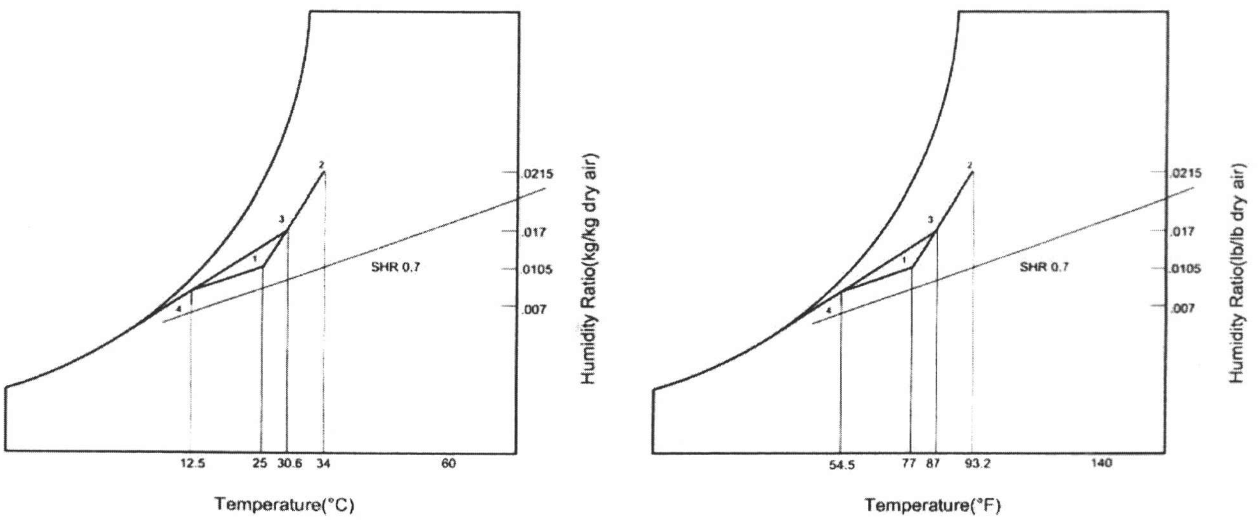

Figure 9b Typical psychrometric representation of a conventional air-conditioning system.

Table 2. Chiller Load for Conventional System (SI)

Required Air Volume* m³/h	% of Ventilation	Ventilation Air m³/h	Recirculated Air m³/h	Chiller Load kW
275	0	0	275	1.70
	10	27.5	247.5	2.03
	20	55	220	2.35
	30	82.5	192.5	2.67
	100	275	0	4.99

* Conventional all-air system

Table 2. Chiller Load for Conventional System (I-P)

Required Air Volume* ft³/h	% of Ventilation	Ventilation Air ft³/h	Recirculated Air ft³/h	Chiller Load Btu/h
9713	0	0	9713	5800
	10	971.3	8742	6926
	20	1943	7770	8018
	30	2914	6799	9110
	100	9713	0	17026

* Conventional all-air system

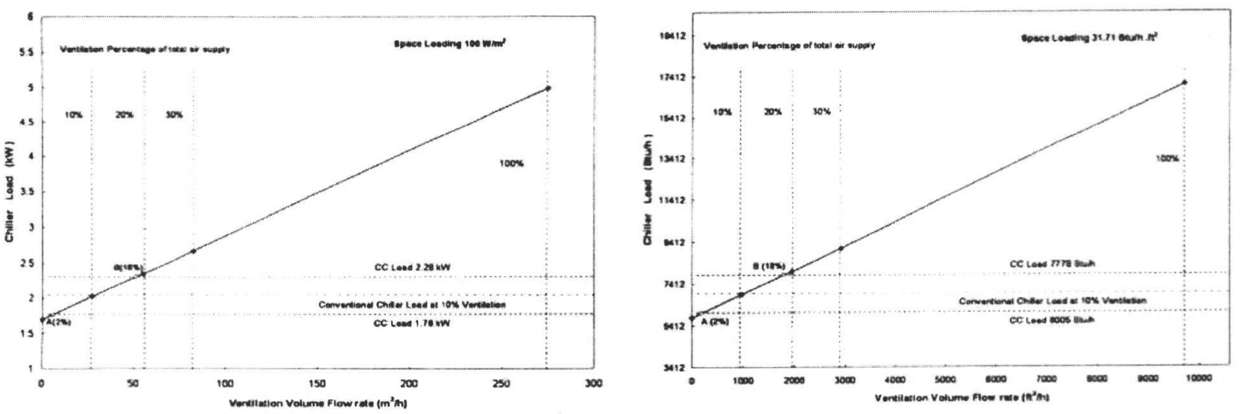

Figure 10 Chiller load vs. ventilation flow rates for conventional air conditioning.

the chiller downsizing threshold of the hybrid system in comparison with the conventional system. The same graph indicates that for the hybrid cycle chiller, downsizing is practicable when the ventilation air requirement is in excess of 2% (point A) for minimum load condition for the hybrid cycle. For the case of maximum hybrid cycle loading, the downsizing threshold is 18%, i.e., whenever the ventilation requirement in a conventional system is greater than 18% (point B), the chiller capacity in the hybrid cycle can be downsized.

Another horizontal line is drawn along chiller load of 2.03 kW (6,926 Btu/h), which is the load for a conventional all-air system with 10% ventilation air and bypass factor of 15%. This line serves as a benchmark chiller load for the purpose of evaluating chiller downsizing potential expressed by the following equation.

$$\% \text{ downsizing} = \frac{\text{Conventional system chiller load} - \text{Chilled ceiling system chiller load}}{\text{Conventional system chiller load}} \times 100$$

From Table 3 and Figure 11 it may be observed that for a very conservative ventilation rate of 10%, the downsizing potential is 13.3%. For the case when ventilation air supply is in accordance with ASHRAE Standard 62 (ASHRAE 2001), i.e., 24.4%, the same rises to 29%. For the case of 100% ventilation air, e.g., operating theater or specific hospital ward, the downsizing potential is as high as 64.7%. This is comparable to the observation of Zhang and Niu (2003b), who cited a figure of 50% energy savings for a similar system. Should the volume flow rate of the hybrid cycle be increased to 250 m^3/h (8,830 ft^3/h), downsizing potential would range between −12.3% for 10% ventilation rate to 54.3% for 100% ventilation rate.

Indicative Energy Consumption

While comparing the performance of this hybrid system with that of a conventional air-conditioning system, one of the main considerations would be the relative energy consumption. The main advantages of hydronic radiant cooling are: (a) chilled water supply at around 13°C (55.4°F) is adequate as against 6°C to 7°C (42.8°F to 44.6°F) for a conventional system, (b) fan power to supply ventilation air is a fraction of that of the conventional system, and (c) pump work for chilled water circulation is about the same. The main disadvantages of desiccant systems are the need for desorption of moisture removed from the supply air and the consequent temperature rise of desiccated air. However, the regeneration process can be achieved by employing low-grade energy at temperatures between 60°C (140°F) and 100°C (212°F). Use of solar energy or waste heat could be an economical option.

Table 3. Downsizing Potential for Hybrid Cycle (SI)

Required Air Volume* m^3/h	% of Ventilation	Chiller Load kW	Downsizing Based on CC Load 1.76 kW (67 m^3/h) %	Downsizing Based on CC Load 2.28 kW (250 m^3/h) %
275	0	1.70	−6	−34
	10	2.03	13.30	−12.31
	20	2.35	25.10	2.97
	24.4	2.49	29	8.08
	30	2.67	34.08	14.60
	100	4.99	64.72	54.30

* Conventional all-air system

Table 3. Downsizing Potential for Hybrid Cycle (I-P)

Required Air Volume* ft^3/h	% of Ventilation	Chiller Load, Btu/h	Downsizing Based on CC Load 6005 Btu/h (2366 ft^3/h) %	Downsizing Based on CC Load 7199 Btu/h (8830 ft^3/h) %
9713	0	5800	−6	−34
	10	6926	13.30	−12.31
	20	8018	25.10	2.97
	24.4	8496	29	8.08
	30	9110	34.08	14.60
	100	17026	64.72	54.30

* Conventional all-air system

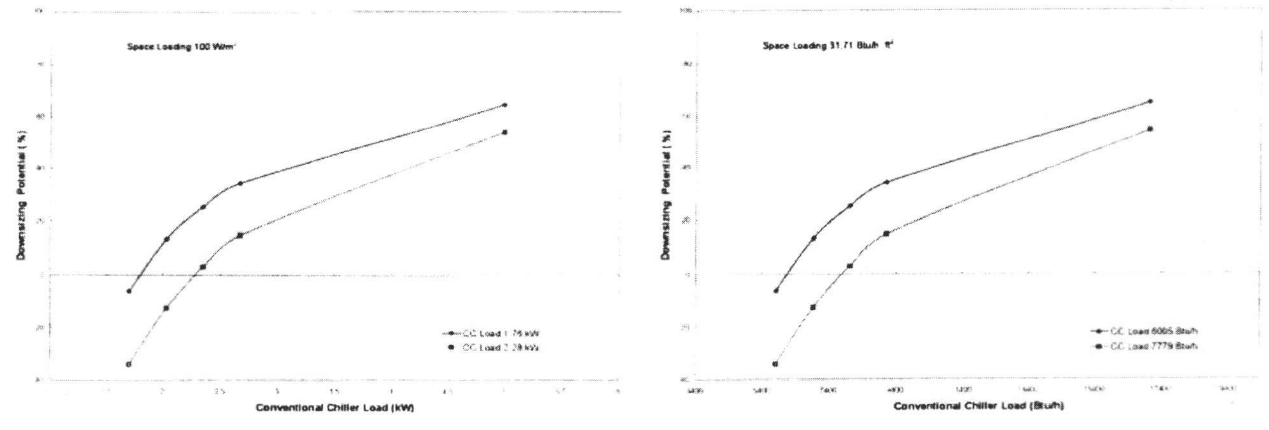

Figure 11 Chiller Downsizing potential for the hybrid cycle.

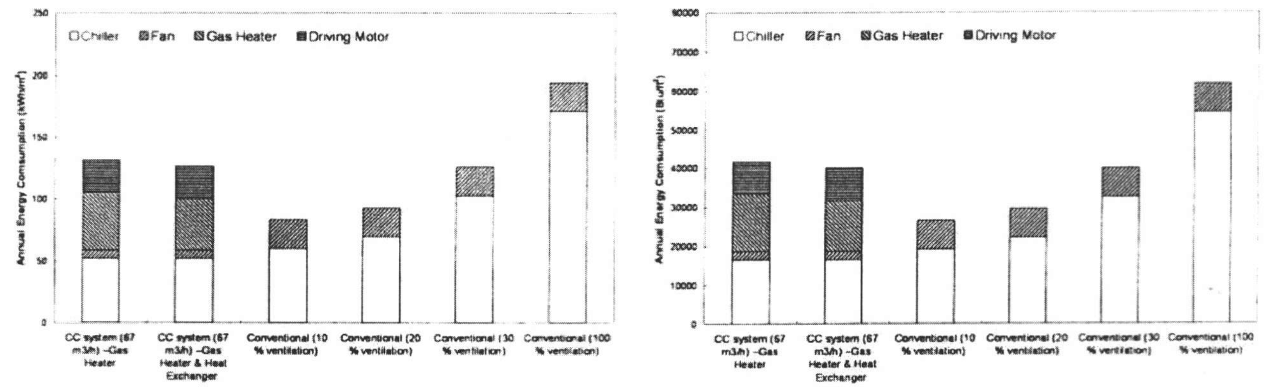

Figure 12 Comparative indicative annual energy consumption of hybrid and conventional systems.

Following the above analysis, an annual energy consumption for the hybrid system has been estimated. In the calculation, where direct heat supply (say, gas) to the desiccant wheel is considered for moisture desorption, equivalent electrical consumption has been divided by a factor of 3. The following were used in the calculation: fan (pump) efficiency, 0.60; fan pressure rise of 1400 Pa (0.203 lbf/in.^{2}) for all-air system and fan pressure rise of 1600 Pa (0.232 lbf/in.^{2}) for desiccated air supply. Chiller COP is 4.39. Desiccant regeneration energy is calculated based on a regeneration temperature of 80°C (176°F), which is similar to that considered by Zhang et al. (2003b). The fan power is calculated from the following equation:

$$\text{Fan power} = \frac{V_a \times \Delta P}{3600\eta_f}(W) \tag{3}$$

where

V_a = air volumetric flow rate,

ΔP = total pressure rise, Pa,

η_f = fan efficiency.

Annual energy consumption has been calculated for the hybrid cycle and has been compared with energy consumption for the all-air system with different ventilation rates. The results are shown in Figure 12. For the CC system where the supply air volume is 67 m^3/h (2366.44 ft^3/h) air, the annual energy consumption is 131.34 kWh/m^2 (41,627 Btu/ft^2) when a gas heater is considered for reactivation of the desiccant wheel (use of an electrical heater raises the consumption to 224.42 kWh/m^2 [71,163 Btu/ft^2]). For the conventional system, consumption ranges from 83.56 kWh/m^2 (26,497 Btu/ft^2) or 10% ventilation air to 194.76 kWh/m^2 (61,758 Btu/ft^2) for 100% ventilation air. For a ventilation air rate of 30% and above, the proposed hybrid system becomes more energy efficient. Note that this ventilation air rate is very close to that recommended by ASHRAE. Furthermore, there is still some room for further energy savings by interposing an additional heat exchanger, as shown in Figures 1 and 4. With the use of the same heat exchanger, energy consumption may be reduced to 126.33 kWh/m^2 (40,059 Btu/ft^2). If solar energy could be harnessed even for partial desorption, further economy can be achieved.

PROJECT STATUS

The basic facility, as described earlier, is in place. Trial runs conducted confirm that even at 15°C (59°F) chilled panel temperature, a condensation or sweating problem did not arise and a comfortable environment could be maintained. With respect to operational economy, certain improvement is warranted. The installed commercial dryer with built-in electrical heater consumes excessive power for reactivation of the desiccant wheel. However, by replacing the electrical heater by a gas heater and interposing an air heat exchanger in the circuit (as shown in Figures 1 and 4), operational economy can be substantially improved, as pointed out earlier. In the modified cycle, the water precooler may also be eliminated.

CONCLUDING REMARKS

The critical challenges for the present hybrid system are (a) whether the total load can be handled economically by the chilled panels in humid tropical climates, (b) whether chilled panels can be operated without condensation, and (c) how its energy consumption compares to that of a conventional air-conditioning system. The trial run confirmed that the system is functional, and the condensation or sweating problem is not insurmountable. For a space load of 0.1 kW/m^2 (31.71 Btu/h·ft^2), any ventilation rate above 2% for a conventional system offers opportunity for downsizing chiller capacity of the hybrid system. Considering a very conservative ventilation rate of 10%, the downsizing potential for chiller capacity is 13.3%. For the case where the ventilation air supply accords with ASHRAE Standard 62 (ASHRAE 2001), i.e., 24.4%, the same rises to 29%, while for the case of 100% ventilation air, e.g., an operating theater or specific hospital ward, the downsizing potential is as high as 64.7%. Based on an indicative energy analysis, the proposed hybrid system becomes more energy efficient than a conventional system when the required ventilation rate is 30% and above.

In conclusion it may be stated that the study indicates definite merit for the hybrid system when the ventilation air requirement of a conventional system is above a certain threshold. This is particularly so in many practical applications where a high ventilation air requirement is essential or mandated, such as operating theaters and certain hospital wards. Apart from functional viability, the study shows that the system could also be economical.

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